5030-537
Electric & Hybrid Vehicle System
Research & Development Project

DOE/CS54209-7
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Environmental Control Subsystems for Electric and Hybrid Vehicles

A Study Evaluation Report

Burton Zeldin

April 15, 1982

Prepared for
U.S. Department of Energy
Through an Agreement with
National Aeronautics and Space Administration
by
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ABSTRACT

Results of parallel contracts awarded by the U.S. Department of Energy's Electric and Hybrid Vehicle Program to TRW and MTI are summarized. Both contractors conducted studies to compare all known heating and cooling mechanisms and methods for the purpose of determining those best suited for the Environmental Control Subsystem (ECS), a nonpropulsion technology for electric and hybrid vehicles. Rationale is presented for recommendations concerning the further development of ECS candidates that are significant departures from current ECS technology. Ranking data and methods are excerpted from the contractor's final reports and are evaluated. Recommendation is given for further study of the split absorption heat pump/refrigerator ECS scheme developed by MTI, leading to a proof-of-concept development.

ACKNOWLEDGMENT

The Electric and Hybrid Vehicle (EHV) Research, Development and Demonstration Act of 1976, Public Law 94-413, later amended by Public Law 95-238, established the governmental EHV policy and the current U.S. Department of Energy (DOE) EHV Program. The EHV System Research and Development Project, an element of this program, is being conducted by the Jet Propulsion Laboratory (JPL) of the California Institute of Technology through an agreement with the National Aeronautics and Space Administration.

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EXECUTIVE SUMMARY

The Environmental Control Subsystem (ECS) is the assemblage of heating and cooling elements and associated hardware that regulates the thermal environments of the passenger space and battery pack in an electric or hybrid vehicle (EHV). The function it performs requires considerable energy in a vehicle whose limited supply of onboard energy is primarily dedicated to vehicle propulsion. The ECS energy demand and even its mass tend to degrade such critical vehicle performance factors as range and acceleration, yet passenger comfort and safety, marketing considerations, and federal regulation create a firm need for the ECS.

Because of obvious differences in the abilities of internal combustion engine-powered vehicles and EHVs to supply large quantities of waste heat, shaft work, etc., uncertainties exist as to whether conventional automotive ECS technology can provide an adequate basis for design of a fully acceptable EHV ECS. However, many alternative heating and cooling elements, i.e., mechanisms such as combustion heaters, air conditioners, heat pumps, etc., and methods in addition to those now in automotive use are known to science. Some of these methods have been used for many years, some have been recently introduced to practice, some are in early stages of development, and some have only recently been discovered. Therefore, a study was sponsored by the U.S. Department of Energy (DOE) Electric and Hybrid Vehicle Program to examine all known heating and cooling mechanisms and methods to determine those best suited for EHVs. The Jet Propulsion Laboratory (JPL) of the California Institute of Technology conducts an EHV System Research and Development Project for the DOE EHV Program, and this Project was given responsibility for guiding and evaluating the ECS study.

Two essentially identical but independent contractor studies were conducted to identify the ECS elements best suited for use in electric and hybrid vehicles. The contractors were Mechanical Technologies Inc. (MTI) in Latham, New York (Contract No. 955682), and TRW Energy Systems Planning Division in McLean, Virginia (Contract No. 955683). The dual award was made to obtain the benefit of more than one contractor's point of view. The Program strategy was to encourage the contractors to independently pursue innovative approaches to the overall problem of environmental control. The contractors were asked to define functional requirements and develop a rating scheme that would eliminate inappropiate elements and lead to the ranking and ultimate identification of two different "best" ECS configurations: (1) based on state-of-the-art technology for possible immediate prototype development, and (2) based on innovative ECS concepts that require more extensive development. The contractors were required to describe the fundamental thermodynamic, electrical, and mechanical principles of operation of each ECS element identified in the study. During the course of the study, a wide variety of alternative ECS heating and cooling elements were evaluated, compared, and ranked. Ultimately, each contractor recommended the ECS configurations it regarded as "best" for EVs and HVs for both the near-term and for more advanced prototype development. The recommendations of the contractors are presented in Table 1. This report presents a rationale for the selection of the split absorption heat pump/refrigerator as a candidate for further development studies.

Table 1. Contractor-Recommended ECS Configurations

Application	MTI	TRW
Near-Term ECS for EV	Gasoline Engine-Driven Vapor Compressor Heat Pump/Refrigerator	Combustion Heater and Gasoline Engine-Driven Vapor Compression Refrigerator
Near-Term ECS for HV	Same as Near-Term ECS for EV	Combustion Heater and Vapor Compression (Refrigerator Driven by Mechanical Drive Output of HV Motor/Engine)
Advanced ECS for EV	Ammonia-Water Split Heat Pump/Refrigerator	Ericsson Engine-Driven (or Electric Motor-Driven) Ericsson Heat Pump/Refrigerator
Advanced ECS for HV	Same as Advanced ECS for EV	Same as Advanced ECS for EV

As can be seen from Table 1, the vapor compression cycle provides the air conditioning capability in each near-term ECS recommended. Virtually all current automotive air conditioners are based on this cycle. Except for TRW's HV version, all recommended near-term ECSs require a separate, small gasoline engine. For heating, MTI suggests operating the same vapor compression system as a heat pump, while TRW recommends the use of a separate combustion heater instead.

While the vapor compression cycle-based systems do not offer a substantial departure from current automotive ECS technology, the advanced systems certainly do. The Ericsson engine-driven Ericsson cycle heat pump/refrigerator is a heat-driven system that, according to TRW, is expected to operate at a very impressive 80 to 90% of Carnot efficiency. However, this system appears to be in a very rudimentary state of development.

The ammonia-water split absorption heat pump/refrigerator recommended by MTI is based on state-of-the-art technology and will probably require a

relatively minor development effort. The basic system provides both heating and cooling, is simple, requires no onboard fuel supply and very little onboard electrical energy.

After considering the advantages and disadvantages of each contractor-recommended ECS and the probable development effort required, the split absorption heat pump/refrigerator has been selected for prototype development. Development would begin with a feasibility experiment leading to the issuance of a Request For Proposal for procurement of a proof-of-concept ECS.

SECTION I

INTRODUCTION AND BACKGROUND

Controlling the thermal environment in an electric or hybrid vehicle presents a significant technical challenge, as the energy required to perform this task is a sizable fraction of the very limited onboard energy supply dedicated primarily to the traction motor. The subsystem that performs the thermal control function has been termed the Environmental Control Subsystem, or ECS.

This report summarizes the results of two essentially identical but independent contractor studies conducted to identify the ECS elements best suited for use in electric and hybrid vehicles. A rationale for the selection of a system as a candidate for further development is constructed from this summary.

The physical make-up of the ECS evolved from a wide variety of environmental requirements and considerations, including some which are not directly associated with driver and passenger comfort. For example, the seasonal potential for windshield icing in certain parts of the United States and the associated reduction in driver visibility motivated the requirement for windshield defrosting. The importance placed on good driver visibility is underscored by the existence of Federal Motor Vehicle Safety Standard No. 103 (Ref. 1), which establishes minimum performance requirements for windshield defrosters in passenger vehicles. Windshield fogging is another weather-related phenomenon which adversely affects driver visibility. It occurs whenever the windshield temperature is lower than the dew point temperature of the air in contact with it. Unlike icing, which occurs only during cold spells (32°F or less), fogging will occur even on warm days if the humidity is sufficiently high. To counteract potential windshield icing and/or fogging, the ECS requires a source of heat and means for delivering it to the windshield.

Another noncomfort consideration stems from the well-established fact that performance of the lead-acid battery, the direct source of power for EV and HV traction motors, exhibits strong temperature dependence. Although the operating temperature that optimizes the trade-offs between vehicle range, battery life-cycle costs, charge rates, etc., has not yet been established, evidence to date indicates that it may be considerably higher than normal ambient temperature and probably in the neighborhood of 120°F (49°C). This elevated temperature suggests the need for a battery compartment heater with temperature control capability in addition to thermal insulation to minimize the rate of heat loss to the ambient environment.

As for driver and passenger comfort, a heater is a necessity during cold winter months in many parts of the United States and is ordinarily supplied as standard equipment in conventional automobiles. The need for a passenger compartment heater might also be argued from the viewpoint of road safety because concentration at the wheel would certainly be diminished if the driver were too cold.

At the opposite extreme is the necessity for passenger compartment cooling during warm summer months. Although an air conditioner is not ordinarily a standard equipment item, there is ample evidence to suggest that the marketability of an EV or HV would be seriously impaired if an air conditioner were not available as an option. Consider, for example, that approximately 80% of the new cars purchased in the United States are equipped with air conditioners. But an air conditioner must be reasonably priced, and neither its presence nor usage should significantly degrade the already limited driving range and performance of an EV or (to a lesser extent) HV.

From all these considerations arose the definition of the ECS for an EV or HV as an affordable assemblage of components with the primary function of providing passenger compartment and battery compartment temperature control, along with required windshield defogging and defrosting capabilities, with only minimal consumption of onboard electrical energy.

It was evident from the start that conventional automotive heating and air conditioning elements would not be practical candidates for use on EVs and HVs because of the differences in characteristics between these vehicles and those powered by an internal combustion engine (ICE). For example, the energy for warming the passenger compartment in conventional cars is derived from heat rejected by the ICE. There is no such equivalent source of a plentiful supply of high temperature waste heat in EVs. Even in the HV presently under development by General Electric Company, the ICE, which could supply considerable waste heat when running, is not in continuous use. Thus the HV ICE cannot be relied upon to supply waste heat to a heat-driven ECS on demand.

The alternative of recovering waste heat from the traction motor or power conditioning equipment presents problems as well. Because the conversion efficiencies of these items are high, the available waste heat is not plentiful, and future trends will be to further reduce such electrical losses. In addition, these losses occur in widely distributed parts which do not operate at high temperature. It is not clear that the economic value of the waste heat that could be saved for use by the ECS would offset the cost of, for example, a fluid loop, heat exchanger, pump, and fan required to accomplish the recovery process.

The foregoing arguments strongly suggested the need for a comprehensive study in which alternative cooling and heating elements would be identified, characterized, described, compared, and rated in an attempt to identify the most suitable ECS elements for prototype development. Because the consensus of opinion was that the expertise and experience in air conditioning and heating required to perform a credible study resides primarily in private industry, a Request For Proposal (RFP) was constructed that outlined the study to be performed. This was advertised in the Commerce Business Daily. About 30 requests were received for inclusion on the source list and, ultimately, five proposals were received and evaluated. In order to obtain more than one point of view, two essentially identical study contracts (Contract Nos. 955682 and 955683) were awarded to the two highest scoring proposers -- Mechanical Technologies Incorporated (MTI), in Latham, New York, and TRW's Energy Systems Planning Group in McLean, Virginia, respectively.

The RFP Statement of Work, shown in Appendix A, provides a detailed description of what was called for in the study. However, the essence of the study is outlined in the following eleven tasks that the contractors were asked to perform:

(1) Develop functional requirements of ECS elements (e.g., heater, air conditioner, battery temperature controller).

(2) Develop a rating scheme.

(3) Identify ECS elements.

(4) Describe principles of operation of the ECS elements identified.

(5) Eliminate inappropriate ECS elements.

(6) Rank remaining ECS elements.

(7) Identify the "best" ECSs for the near-term and for more advanced development for EVs.

(8) Determine the impact of adding cooling capability to the battery temperature controller.

(9) Repeat Task (7) for HVs.

(10) Estimate cost, schedule, etc., for ECS prototype development.

(11) Document specified tasks and present study conclusions.

SECTION II

CONTRACTOR STUDY RESULTS

A. FUNCTIONAL REQUIREMENTS

As anticipated, there were substantial differences between the study approaches employed by the two contractors. For example, in developing the functional requirements, TRW elected to use analytical techniques for estimating the required heating, cooling, windshield defrosting, and windshield defogging capacities for the ECS. These capacities were required for ECS component sizing and energy demand, among other things. Their thermal analyses included the effects of convection, conduction, radiation, air exchange rates, vehicle glass area, insolation, ambient temperature, relative humidity, wind velocity, desired passenger compartment temperature, and many other factors. MTI, on the other hand, merely assumed the required capacities based on scaling the requirements of similar vehicles uncovered in their literature search of present automotive heating and cooling practices. These differing approaches naturally led to somewhat different design point capacity requirements. However, a given contractor's relative ranking of ECS elements would not be expected to be strongly influenced by design capacity, as the parameters which influence rating score (e.g., mass, volume, energy consumption) vary in roughly the same way as a function of capacity for all elements considered.

The design point ambient temperature range of -20 to 120°F (-28.9 to 49°C) suggested by JPL in the RFP was adopted by TRW with some reservations about the extremities of the range. These reservations were later shown to be justified by an MTI statistical analysis which concluded that ambient temperature will fall outside the range -10 to 100°F (-23.3 to 37.8°C) less than 1% of the time for less than 1% of all cars in the United States. Their study was based on car registration and weather data for the 25 largest standard metropolitan statistical areas in the United States. A similar statistical analysis by MTI established the maximum design point insolation level of 326 Btu/h ft^2 (1028.4 W/m^2). TRW assumed 300 Btu/h-ft^2 (946 W/m^2) based on their literature study.

The MTI and TRW functional requirements and design points are partially summarized in Exhibits 2-1a and 2-1b, derived from their final reports (Refs. 2 and 3).

B. RATING SCHEMES

Significant differences exist between the rating schemes developed by the contractors, even though both are derivatives of the one suggested as an example in the RFP.

Design Load Specifications:

- Continuous 17,000 Btu/hr (≈5 kW)
- 2.5 hours of operation maximum (passenger compartment)
- 42,500 Btu (maximum)
- 10 hours of recharging
- 10 hours of unplugged operation for battery temperature controller.

Passenger Compartment Design Point Specifications:

Conditions	For Heating Season	For Cooling Season
T_a	-10°F	100°F
T_w	-	74°F
Air Velocity	45 mph	45 mph
Solar Insolation	-	326 Btu/hr/ft^2

Ambient Condition Design Point Specifications:

Air Exchange > 5 cfm/person		
Parameter	During Heating Season	During Cooling Season
T_a, dry-bulb temp.	>68°F	<75°F
T_w, wet-bulb temp.	-	<75°F
Air velocity at the passenger	<0.5 meter/sec	<1.5 meter/sec
T_{mr}, mean radiant temp.	Limit Not Specified	

Design Point Fresh Air Exchange:

- For electric vehicles utilizing ECS elements which will not result in any hazardous fumes or gases - 5 ft^3/min/person
- For electric vehicles utilizing ECS elements which may result in the generation of hazardous fumes or gases - 15 ft^3/min/person
- For hybrid vehicles - 15 ft^3/min/person.

Exhibit 2-1a. Functional Requirements, MTI (Ref. 2)

Summary of Functional Requirements for Integrated Environmental Control System

Heating Requirements

1. Capable of maintaining a temperature of at least 18°C (65°F) in passenger compartment at -29°C (-20°F) ambient.
2. Heating capacity 5.7 KW (19,000 Btu/hr) with full ventilation load. Design range is 2.2 to 2.8 KW (7,500 to 9,500 Btu/hr) with controlled ventilation.
3. Time to reach full capacity is less than 10 minutes.
4. Maximum air flow capacity of 1100 m^3/hr (200 cfm) for defroster.
5. Heat can be directed to passenger compartment, defroster jets, or battery compartment.
6. Optional: embedded electrical heat source of 0.6 KW in windshield.
7. Heater can recover battery from "cold soak" in less than 4 hours.

Cooling Requirements

1. Capable of maintaining a temperature of 25°C (77°F-RH 40%) in passenger compartment at 49°C (120°F) (29°C (85°F) wet bulb) ambient.
2. Cooling capacity 7.5 KW (25,500 Btu/hr) with full ventilation load. Design range is 3.2 to 4.1 KW (11,000 to 14,000 Btu/hr) with controlled ventilation.
3. Time to reach capacity less than 3 minutes.
4. Time to produce comfort for front seat passengers less than 10 minutes.
5. Optional: cooling air stream can be directed to battery compartment.

General Requirements

1. Capable of withstanding mechanical, thermal, vibrational, and acceleration environment of electric or hybrid vehicles.
2. System completely sealed and designed so combustion air and exhaust products do not enter passenger compartment.
3. Minimizes use of hazardous materials which could be released in normal operation or accidents.
4. Capable of being packaged and integrated into the vehicle system.

Battery Temperature Controller Functional Requirements

Assumptions

1. Batteries as specified in Reference 2-16 (FL).
2. Battery packaged widthwise in vehicle. Thermal conductance of container to ambient 4.9 watts/°C (9.3 Btu/hr°F).

Controller Requirements

1.	Minimum Battery Temperature (Operating mode - no heat supplied in cold soak mode)	= 18°C (65°F)
2.	Maximum Battery Temperature (Maximum cooling demanded at this temperature - Separate requirements will be developed in Section 7.1)	= 49°C (120°F)

Heating and Cooling Requirements

1.	Maximum Heat Required to Maintain Operating Temperature at 49°C (120°F).	= 380 watts (1295 Btu/hr)
2.	Minimum Heat Energy Required to Recover from Cold Soak Conditions	= 7.0 kWh (24,000 Btu)
3.	Maximum Heat Removal Rate (Based on 8 hour charging cycle)	= 225 watts (767 Btu/hr)
4.	Cooling Fan Flow Required to Remove Charging Heat Release (Based on 6°C (10°F) air temperature rise and 8 hour charge period)	= 120 m^3/hr (70 cfm)

Exhibit 2-1b. Functional Requirements, TRW (Ref. 3)

MTI adopted the relationship,

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$$S = \sum_{i=1}^{N} W_i \quad s_i / \sum_{i=1}^{N} W_i$$

Where: S = Total system or element score, nondimensional.

W_i = Weighting factor assigned to the i^{th} criterion (rating factor) to indicate its relative importance to other criteria, nondimensional.

s_i = Rating score of the i^{th} criterion (rating factor), nondimensional.

In turn, s_i is given by either

$$s_i = \begin{cases} (C_i)_{ref}/C_i \text{ (for Type I criteria), or} \\ C_i/(C_i)_{ref} \text{ (for Type II criteria)} \end{cases}$$

depending upon whether small or large values of the i^{th} criterion C_i are more desirable, respectively. The quantity $(C_i)_{ref}$ is the value of the i^{th} criteria for a selected reference systems to which all others are compared. Examples of Type I criteria are cost and weight, whereas lifetime and energy efficiency are Type II criteria.

A summary of the specific criteria and weighting factors used by MTI and TRW are categorized and listed in Exhibits 2-2a and 2-2b.

It is of interest to note that the definition of s_i adopted by MTI permits the existence of elements with one or more s_i greater than unity. Because the sum of the weighted factors is 100, not only can s_i exceed 100, but it has no upper bound. This, in and of itself, is of minor consequence because the element with the highest total score S could still be judged "best." However, the subtle danger is that an element which ranks very highly in one characteristic and as poorly as zero in the others could conceivably receive the highest total score. Other than engineering judgement, the only barrier to such an occurance is the task which "weeds out" such poor candidate elements and eliminates them prior to any attempt to rate them.

The basic rating scheme equation used by TRW is, on the surface, identical to that used by MTI. However, the rating factors selected (i.e., criteria), their number, and the weights assigned to them are quite different, as shown in Exhibit 2-2b.

It should be observed that all of TRW's rating factors are of Type I (i.e., small values are desired). Moreover, TRW's weighting factors sum to unity and their s_i ranges between zero (worst score) and 100 (perfect score). Thus, S also ranges between zero and 100. TRW elected to separate their rating factors into two categories, which they refer to as Type A and

CRITERIA AND ASSOCIATED WEIGHTING FACTORS

Criterion	Weighting Factor (W_i)
Capital Cost Characteristics:	
1. First Cost	25
2. System Life	5
Use Characteristics:	
3. Range Impact	10
4. Energy Efficiency	5
5. Storage Period	5
6. Maintainance Cost	10
7. Performance Impact	10
Environmental and Safety Characteristics:	
8. Consumer Perception of Safety	5
9. System Noise	10
10. Other Environmental Impacts	5
Development and Manufacturing Characteristics:	
11. Ease of Packaging and Volume	5
12. Development Cycle Through Commercialization	5
TOTAL	100

Exhibit 2-2a. Criteria and Associated Weighting Factors, MTI (Ref. 2)

Weights Selected for Rating Scheme

Rating Factor	Weight (Normalized)	
Cost (Initial)	0.60	
Impact on Vehicle Range		
- Weight Factor	0.13	0.40
- Volume Factor	0.13	
- Energy Use Factor	0.14	
Total	1.00	

Exhibit 2-2b. Criteria and Associated Weighting Factors, TRW (Ref. 3)

Type B. A Type-A factor (Exhibit 2-3) receives a perfect score of 100 if its value is equal to or less than its preselected baseline value. The score decreases linearly if its value exceeds that of its baseline. A score of zero is given if the factor value exceeds its baseline value by a preselected multiple (e.g., 2 or 3). Cost was the only factor classified as Type A. Weight, volume, and energy use were classified as Type B. As illustrated in Exhibit 2-4, the perfect score is awarded only if the value of the factor is zero. The score decreases linearly toward zero as the value of the factor increases. The rate of decrease is determined by the arbitrary score of 25 assigned to a factor with a value equal to its preselected baseline value.

C. ELEMENTS CONSIDERED

Both MTI and TRW identified and listed a large number of potential ECS elements. As would be expected, many elements were common to both lists. However, each contractor identified some elements not listed by the other. In order to conveniently illustrate which elements were identified and by whom, and which were eliminated and by whom, Table 2-1 was constructed based on the contractor final reports (Refs. 2 and 3). Reference should be made to these reports for specific details or descriptions.

D. RANKING

After application of their respective rating schemes, the contractors arrived at the element ordering illustrated in Tables 2-2a and 2-2b. Because there is little commonality between elements ranked in these two tables, and because the ranking schemes are quite different, there is no clear method to compare the MTI and TRW numerical scores relative to the benefits of any one particular element. Still, it is obvious that the one element in common, the gasoline engine-driven vapor compression cycle, was placed at the bottom of the list of elements ranked by type by both contractors. Another interesting observation is that the two Ericsson cycle heat pumps received by far the highest (and nearly equal) scores by TRW, even though the Ericsson heat engine-driven Ericsson heat pump had only a 2% range penalty, while the electric motor-driven Ericsson heat pump range penalty was reported to be more than an order of magnitude higher!

E. CONFIGURATIONS RECOMMENDED BY MTI AND TRW

The thrust of the entire study was aimed at identifying the elements which comprise the most promising ECSs for near-term and more advanced development for both electric and hybrid vehicles. Based on their studies, the contractors selected the ECS configurations shown in Table 2-3 as their recommended choices for prototype development.

The first choice of both contractors for the air conditioner portion of the near-term ECS for the EV is the vapor compression refrigeration cycle powered by a small auxiliary gasoline engine. This choice was made in spite of the relatively low scores given to the vapor compression cycle in Tables 2-2a and 2-2b. TRW argued that even though the ROVAC-based, gasoline engine-driven air conditioner scored higher than the vapor compression cycle

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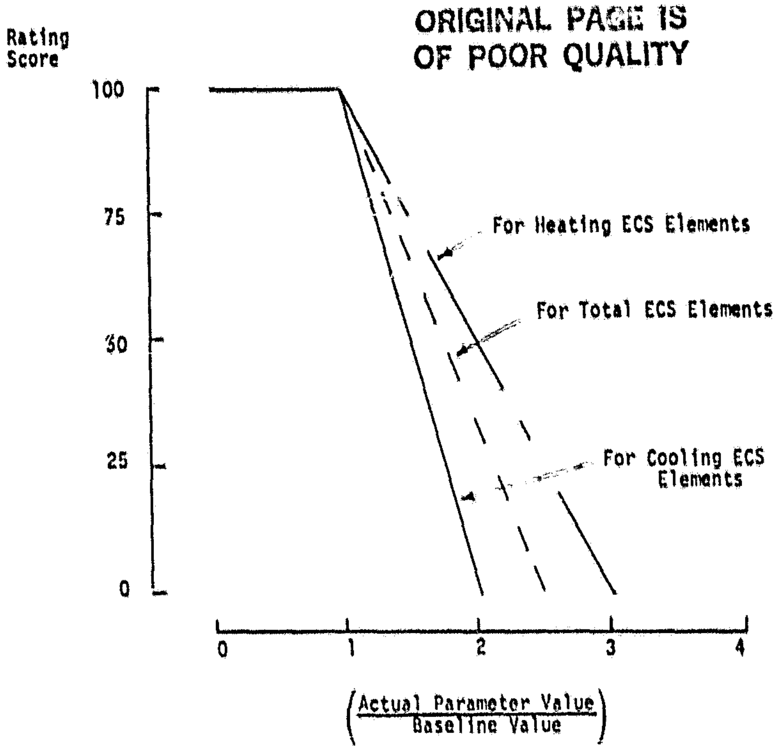

Exhibit 2-3. TRW Type A Functional Form for Cost Parameters (Ref. 3)

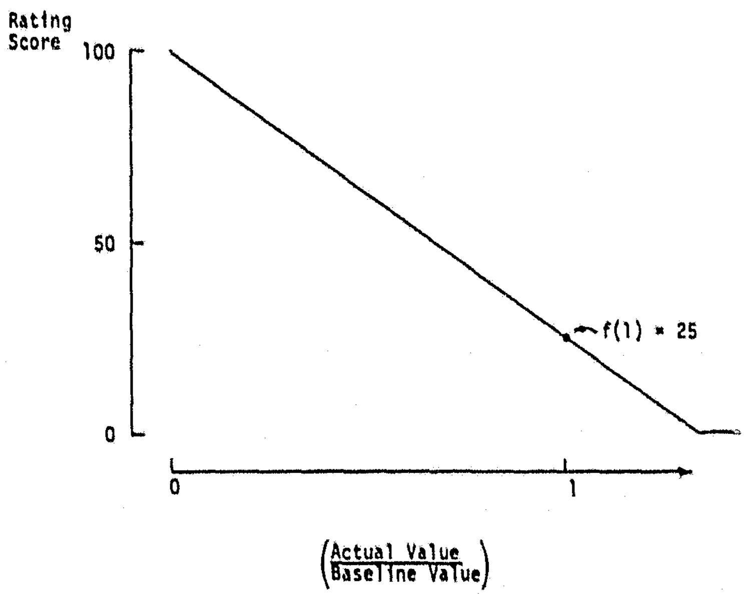

Exhibit 2-4. TRW Type B Functional Form for Weight, Volume, and Energy Use Parameters (Ref. 3)

Table 2-1. ECS Elements Considered by MTI and TRW

Element	Application: H = Heating C = Cooling	Identified By	Eliminated By
Electric Motor-Driven Heat Pump Cycles			
Vapor Compression	H/C	Both	TRW
ROVAC (Reversed Open Brayton)	H/C	TRW	TRW
Ericsson	H/C	TRW	TRW
Electrically-Driven Heat Pump Cycles			
Thermoelectric	H/C	MTI	MTI
Heat Engine-Driven Heat Pump Cycles			
Gasoline (Otto) Engine-Driven Vapor Compression	H/C	Both	-
Gasoline (Otto) Engine-Driven ROVAC	H/C	TRW	-
Stirling Engine-Driven Vapor Compression	H/C	TRW	-
Free Piston Stirling Engine-Driven Vapor Compression	H/C	MTI	-
Ericsson Engine-Driven Ericsson	H/C	TRW	-
Heat-Driven Heat Pump Cycles			
Absorption	H/C	Both	MTI
Hydride	H/C	Both	Both[a]
Vapor Jet Compression	H/C	TRW	TRW
Miscellaneous Heat Pump Cycles			
Split Absorption	H/C	MTI	-
Intermittent Absorption	H/C	TRW	TRW
Magnetic	H/C	MTI	MTI
Direct Conversion Elements			
Combustion Heaters	H	TRW	-
Resistance Heaters	H	TRW	TRW
Thermal Energy Storage (TES) Elements			
Water and Water-Based Solutions	H/C	Both	-
Polyethylene Pellets	H	MTI	MTI
Alkali Carbonates	H	MTI	-
Hydroxides	H	MTI	-
NaOH - $NaCO_3$	H	MTI	-
LiF	H	Both	-
Phase Change "Salts"	H	MTI	MTI
Organic Oils	H	MTI	-
Paraffins	H	MTI	MTI
Sand	H	MTI	MTI
Expendable Liquified Gases	C	Both	Both
TES Cores	H	TRW	TRW
Others			
Compressed Air	H/C	MTI	MTI
Reversible Thermochemical Reactions	H/C	Both	Both
Evaporative Cooling	C	TRW	TRW

[a]MTI eliminated all hydrides considered except MgHx.

Table 2-2a. MTI Ranking

Element Function	Element	Rank	Score	Range Penalty
Heating	Water Thermal Energy Storage (TES)	1	377	14%
	K_2CO_3-Li_2CO_3 TES	2	208	10%
	Li_2CO_3 TES	2	208	10%
	Organic Oil TES	3	198	15%
	LiOH TES	4	195	14%
Cooling	LiBr Water Split Absorption	1	213	8%
Heating and Cooling	Water Thermal Energy Storage	1	240	14%
	Aqua-Ammonia Split Absorption	2	195	12%
	Free Piston Stirling-Driven Vapor Compressor	3	136	11%
	Gasoline (Otto) Engine-Driven Vapor Compressor	4	100	8%

Table 2-2b. TRW Ranking

Element Function	Element	Rank	Score	Range Penalty
Heating	Combustion Heater	1	Not given	Not given
Cooling	Gasoline (Otto) Engine-Driven ROVAC	1	63-72	2-4%
	Ice Making	2	55-64	8-10%
	Gasoline (Otto) Engine-Driven Vapor Compression	3	47-59	3-5%
Heating and Cooling	Electric Motor-Driven Ericsson	1	87	21%[a]
	Ericsson Engine-Driven Ericsson	2	86	2%

[a]The range penalty for this element exceeds the abitrarily set 20% limit stated in the RFP.

Table 2-3. Contractor-Recommended ECS Configurations

Application	MTI	TRW
Near-Term ECS for EV	Gasoline Engine-Driven Vapor Compressor Heat Pump/Refrigerator	Combustion Heater Gasoline Engine-Driven Vapor Compression Refrigerator
Near-Term ECS for HV	Same as Near-Term ECS for EV	Combustion Heater Vapor Compression Refrigerator-Driven by Mechanical Drive Output of HV Motor/Engine
Advanced ECS for EV	Ammonia-Water Split Heat Pump/Refrigerator	Ericsson Engine-Driven (or Electric Motor Driven) Ericsson Heat Pump/Refrigerator
Advanced ECS for HV	Same as Advanced ECS for EV	Same as Advanced ECS for EV

version, the degree of its superiority did not appear to warrant the higher risk and capital investment required for its development and production -- especially considering the small size of the anticipated buyer's market. Another factor that biased TRW in favor of the vapor compression cycle is that this cycle is already in widespread automotive use. MTI reasoned that their recommended vapor compression system is based on state-of-the-art technology and would therefore require the least development time and cost.

Whereas TRW recommends using a combustion heater to provide the heating function for the near-term ECS, MTI suggests using the same gasoline engine-driven vapor compression system as a heat pump. The MTI recommendation offers the advantage of eliminating the need for a separate heater and in addition would use less fuel for the same heat output, as the heating coefficient of performance would exceed unity. A potential disadvantage is that the same ECS element would be required to endure the extra wear and tear of providing both heating and cooling functions. The hardware required to switch between heating and cooling modes may also be a moderate cost factor. Both contractor-suggested systems appear technically feasible because they utilize only state-of-the-art technology. However, while it is tempting to speculate

that a gasoline engine of the type used to power a lawn mower might, after slight modification, be suitable for ECS application, a more careful consideration of some of the desired refinements suggests otherwise. For example, the need must be considered for:

(1) Extended engine life.
(2) Automatic starting capability.
(3) Automatic choking.
(4) Noise and vibration minimization.
(5) An exhaust management system.
(6) An engine cooling system.

MTI recommends the same near-term ECS for both EVs and HVs, whereas TRW recommends modifying their near-term EV ECS for HV application by driving the vapor compressor with mechanical power derived from the HV motor/engine. This eliminates the need for the separate gasoline engine. It should be pointed out that the air conditioner concept recommended by TRW for the HV is virtually identical to the one adapted into the design of the GE Hybrid Test Vehicle. By combining the ideas of TRW and MTI, one could envision a motor/engine-driven vapor compressor used as refrigerator or heat pump depending upon the need at the time. While use of the HV motor/engine to drive the ECS offers the distinct advantage of eliminating the need for and the potential problems associated with a separate gasoline engine, several disadvantages are readily apparent. First, it is obvious that any demand for power by the ECS will decrease the power available for traction. Those who drive a conventional auto equipped with a small engine know that the loss of performance associated with the use of the air conditioner, especially while driving uphill, is far from insignificant. A more subtle disadvantage is that the rotational speed of the vapor compressor will be clamped to that of the motor/engine (unless a variable speed coupling is used). Thus the compressor cannot, in general, be operated at the optimum speed for its thermal loading. The ECS energy efficiency is bound to suffer as a consequence.

TRW recommends the same advanced ECS for both the EV and HV -- an Ericsson cycle heat pump/refrigerator, driven by an Ericsson cycle heat engine with helium as the working fluid.[1] The heat engine would obtain heat from the combustion of a fuel such as gasoline. If, however, high energy density batteries become available, TRW recommends considering using an electric motor instead. Because their own data indicates that putting this system into use would result in a vehicle range reduction of 21 to 27% as opposed to 2% for the Ericsson heat engine-driven version, battery energy densities would have to improve drastically before such a system would become acceptable. According to TRW, the Ericsson heat pump is expected to operate at a very impressive 80 to 90% of Carnot efficiency and is currently under development.

MTI recommends what it refers to as a water-ammonia split heat pump for the advanced ECS for both the EV and HV. This system derives its name from the fact that the ammonia-water absorption cycle is split into two parts.

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[1] Appendix B provides a description of the Ericsson-Ericsson heat pump/refrigerator, excerpted from Reference 4.

The segment of the cycle performed in the vehicle (Figure 2-1) starts with a charge of liquid ammonia in a refrigerant tank and a charge of water in an absorber tank. The pressures between the two tanks are separated by an expansion valve. Liquid ammonia, maintained at a higher pressure than the vapor pressure in the absorber tank, undergoes adiabatic expansion in passing through the expansion valve and in the process is partially vaporized. Because the heat of vaporization is extracted from the ammonia flow itself, the flow becomes quite cold. It then passes through an evaporator where cold liquid ammonia evaporates by absorbing heat. Warm air blown across the evaporator/heat exchanger would be cooled. As the ammonia vapor is absorbed in the absorber tank, heat is released. Cool air blown over the absorbent cooler through which the absorbent is circulated would be warmed. To obtain the air conditioning effect, passenger compartment air is blown across the evaporator and ambient air is blown across the absorbent cooler. To achieve the heating effect instead, the passenger compartment and ambient air flows are diverted to the absorbent cooler and to the evaporator, respectively.

As the concentration of ammonia in the absorber tank increases, so does its vapor pressure. However, the system would be designed to maintain a sufficient pressure difference between the refrigerant and absorber tanks.

The part of the cycle performed external to the vehicle separates the ammonia-water mixture into its components by a distillation process using fixed-place equipment and restores the original charges. This process would presumably take place in the garage at home while the propulsion batteries are being charged. No onboard consumable fuels and very little onboard electrical energy (for control and monitoring functions and for circulation fans) is required. Producing a prototype would probably require a relatively minor effort, as it seems that no high technology would be required for development.

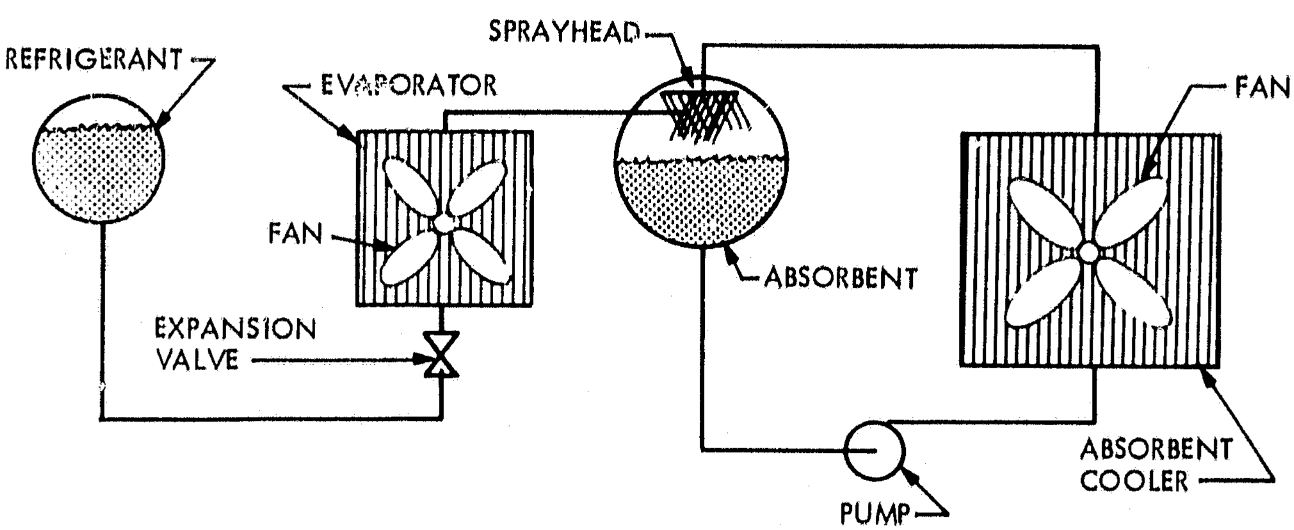

Figure 2-1. Split Absorption Heat Pump/Refrigerator

As an intermediate step, MTI suggests its water Thermal Energy Storage (TES) system, which stores a charge of either hot, high pressure water for heating on cold days or frozen water for air conditioning on hot days. In addition to the obvious disadvantage that any TES system will thermally "self-discharge" at a rate dependent on how well it is insulated, there is the additional subtle disadvantage that the consumer must decide in advance whether a heating or cooling charge will be required. Ordinarily, this would present no problem. However, there are areas in the United States where the ambient temperature sometimes falls well below the comfort zone during evening hours -- in which case some heating capability would be welcomed -- and yet mid-afternoon temperatures soar as high as 90°F or higher. In such areas, the water TES system described in the MTI final report would probably prove to be unsatisfactory.

F. BATTERY PACK TEMPERATURE CONTROL

The battery pack temperature control strategy incorporates the assumption that the pack starts out with a preselected temperature of 120°F (49°C) at the end of each battery charge cycle. The energy required for prewarming would be at least partially derived from internal generation produced during charging. An average temperature rise of somewhat less than 59°F (15°C) could be achieved due to this effect alone even if the batteries were perfectly insulated (based on an estimated maximum heat release of 120 W-h per battery during equilization charging).

The contractors were asked to consider control schemes which would maintain the battery pack at 120°F $\pm$ 10°F (48.9 $\pm$ -5.6°C), except during periods of (vehicle) non-use exceeding 3 days while subjected to a worst case ambient temperature of -20°F (-28.9°C). In addition, they were asked to suggest cooling techniques, should they be required, for limiting the maximum battery pack temperature to 130°F (54.4°C). After considering several control schemes, MTI recommended housing the battery pack in a box-like container made of a simple insulation. It was suggested that a gap be maintained between the inside container walls and the batteries to act as a ventilation space. A small thermostatically-operated fan would blow ambient air through the gap for cooling, if needed. Presumably, the fan could be used for venting battery-produced gases as well. MTI computed the battery pack electrolyte temperature decay with time for worst case ambient conditions. They concluded that their recommended battery pack temperature controller with R-14 insulation [i.e., thermal resistance X area = 14°F-ft^2-h/Btu (2.47°C-m^2/W)] using 2-inch (5.08-cm) thick polyurethane foam is clearly incapable of limiting the minimum battery pack temperature to 110°F (43°C) during the 36 hours of non-use required by the RFP unless heat is supplied. They further indicate that if the propulsion battery pack were used to supply this heat, an unacceptably high fraction of its total energy would be depleted in the process.

The battery pack temperature controller recommended by TRW is conceptually the same as that recommended by MTI. TRW's "strawman" design includes a 1-inch (2.54-cm) thick mineral fiber insulation with a 1/2-inch (1.27-cm) air gap and provisions for diverting heat from their combustion heater to the battery compartment. According to their computations, the

battery pack temperature controller requires about 10% of the design output of the combustion heater. A blower would circulate ambient air for cooling, if required, and would provide a means for positive ventilation. In the unlikely event that the ambient air is not cold enough to provide adequate battery pack cooling, TRW suggests diverting some of the output of the passenger compartment air conditioner to provide this function.

G. REDUCTION OF ECS AIR CONDITIONING LOAD

The strong influence of vehicle design, in particular on the ECS air conditioning load, was recognized by both contractors. For example, their studies clearly asserted that only a small fraction of the fresh air exchange ordinarly provided is required for safe breathing. Because the majority of this flow is used to limit the buildup of odors, and partial removal of its sensible and latent heat accounts for a significant portion of the total cooling load, it makes good sense to reduce the fresh air exchange flowrate to a minimum. TRW estimates that the use of activated charcoal for cleansing the passenger compartment air could reduce the fresh air exchange required to only 15 to 30% of the total ventilation flow. An analysis by MTI showed that a fresh air exchange rate of only 0.5 cfm (2.4×10^{-3} m^3/s) per person would suffice to limit the CO_2 level in the passenger space to a safe level, whereas American Society of Heating, Refrigerating, and Air Conditioning Engineers Building Standard 62-73 for Natural and Mechanical Ventilation suggests 5 to 15 cfm (2.4×10^{-3} to 7.1×10^{-4} m/s) per person. TRW points out that only 1 to 2 lb (0.45 to 0.9 kg) of activated charcoal is needed per person per year in typical building ventilation applications. They go on to project a potential 50% reduction in required ECS cooling capacity if fresh air exchange can be minimized by the use of activated charcoal filters. Supporting this claim is MTI's estimate that more than 40% of the thermal load is presented by infiltration of outside air, either coming in through cracks or leaks or intentionally introduced by blowers.

Most of the other techniques considered for reduction of the air conditioner load focus on methods aimed at reducing solar transmission through vehicle windows. For example, use of louvered sun shields on rear windows, covering selected windows with partially reflective films, and use of photochromic glass have been suggested.

H. ESTIMATES OF PROTOTYPE DEVELOPMENT EFFORT

TRW estimates that prototype development of their gasoline engine-driven vapor compression cycle refrigerator and combustion heater ECS would require a 1-year program costing about $250,000 in labor, not including components, materials, and test facilities.

According to MTI, developing their ammonia-water split heat pump/refrigerator ECS would require about 12,100 man-hours and $161,000 in materials.

SECTION III

CONCLUSIONS AND RECOMMENDATIONS

A. SUGGESTED CONTINUATION FOR ECS PROTOTYPE DEVELOPMENT

A large number of interesting ECS elements were identified, screened, and ranked in an effort to identify the "best" ECSs for near-term and more advanced development. However, a prior program decision restricted consideration for prototype development to those ECS concepts regarded as significant departures from contemporary ECS technology. Of those ECSs ranked "best," only two fall into this category -- the Ericsson-Ericsson heat pump/refrigerator, identified by TRW, and the ammonia-water split heat pump/refrigerator, identified by MTI. Resource limitations prevent a parallel prototype development effort of both systems, so a choice was necessary. While the heat-driven Ericsson-Ericsson ECS offers great potential, it appears to be at a very rudimentary state of development. Prototype development for this system would likely entail high risk and require a long-term effort with commensurately high development costs. On the other hand, although the split heat pump per se has not been developed at all, the hardware and technology required to produce a prototype are well understood, as full-cycle ammonia-water absorption systems have been used in non-automotive applications for quite some time.

Assuming that laboratory testing confirms its viability, the split cycle will undoubtedly require considerably less development than the Ericsson-Ericsson cycle. For these reasons, the split heat pump/refrigerator is recommended for proof-of-concept development. However, in addition to the ammonia-water combination recommended by MTI, other refrigerant-absorbent combinations should be evaluated to determine the best working fluids for the prototype ECS.

In addition, if resources permit, techniques for reducing ECS air conditioner loads should be investigated and, if warranted, verified by testing.

The battery compartment temperature control schemes offered by both contractors rely on insulating the battery pack and circulating warm or cool air in the gap between the batteries and insulation as required. Although this concept appears workable in theory, insulating the battery compartment may be easier said than done because of required insulation penetrations. In any event, because the battery pack will be indifferent to the type of mechanism heating or cooling the air being circulated, it is recommended that the insulated container be developed independently of the heat pump/refrigerator components of the ECS and integrated at a later time.

SECTION IV

REFERENCES

1. Federal Motor Vehicle Safety Standard No. 103-33 FR 6469, April 27, 1968 (Revised July 28, 1975).

2. Bhate, S.K., "Electric and Hybrid Vehicles Environmental Control Subsystem Study," Mechanical Technology Inc., Latham, NY (MTI 81TR36) May 15, 1981.

3. Heitner, K.L., "Electric and Hybrid Vehicle Environmental Control Subsystems Study," TRW-ESPD, McLean, VA (TRW 97649-E005-UX-02) Dec. 4, 1980.

4. Nissen, M.J., and Heitner, K.L., "Electric and Hybrid Vehicle Environmental Control Subsystems - Task 4 Approval Report," TRW-ESPD, McLean, VA (TRW 97649-E003-UX-01) Dec. 2, 1980.

APPENDIX A

EXHIBIT: REQUEST FOR PROPOSAL STATEMENT OF WORK
(Jet Propulsion Laboratory)

SCHEDULE

ARTICLE 1. STATEMENT OF WORK

(a) The Contractor shall perform a study of Environmental Control Subsystems (ECS's) for Electric and Hybrid Vehicles (EHV's). For the purpose of this study, the ECS shall consist of the following three elements: a passenger compartment air conditioner, a passenger compartment heater, and a battery pack temperature controller. Ducting, sensors, control accessories and other necessary items, while not considered elements, are nevertheless part of the ECS.

The major objectives of this study shall be to provide the basis for selection and to select, for the purpose of potential prototype development, those elements comprising the Environmental Control Subsystems best matched to the unique characteristics and requirements of EHV's. Another objective is to generate cost estimates for the development, fabrication, and testing of the ECS systems which will be identified as a result of achieving the major objectives.

In performance of this effort the Contractor is encouraged to consider innovative approaches to the overall problem of environmental control which may offer advantages over more conventional techniques.

In accomplishing the above objectives, the Contractor shall perform the below-delineated tasks using the applicable attachments to Exhibit I.

(1) Task 1 - Development of Functional Requirements

(A) Develop functional requirements for passenger compartment air conditioners, passenger compartment heaters and battery pack temperature controllers individually, and for the ECS as a whole.

(i) In developing functional requirements for air conditioners, construct going to work and going shopping scenarios for Electric and Hybrid vehicles.

(ii) Use Attachments A and B, Exhibit I as guidelines.

(B) Submit the Functional Requirements List to JPL for review and approval.

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(2) Task 2 - Development of a Rating Scheme

(A) Develop a rating scheme and corresponding set of weighting factors for each ECS element individually and for the ECS as a whole.

(i) Use Attachments A and C, Exhibit I as guidelines.

(B) Submit the Rating Scheme to JPL for review and approval.

(3) Task 3 - Identification of ECS Elements

(A) Develop an exhaustive list of candidate ECS elements with a potential to fulfill the functional requirements established in Task (a)(1).

(i) Consider both state-of-the-art technology and innovative ECS element and system configuration concepts.

(4) Task 4 - Description of Principles of Operation of ECS Elements

(A) Provide written descriptions of the fundamental thermodynamic, electrical, and mechanical principles of operation of each ECS element identified in Task (a)(3).

(i) Provide illustrations, state diagrams (when appropriate), and schematics in sufficient detail to facilitate JPL's understanding of the narrative descriptions.

(5) Task 5 - Elimination of Inappropriate ECS Elements

(A) Eliminate those ECS elements identified in Task (a)(3) which cannot meet the functional requirements as identified in Task (a)(1) or which possess inherent qualities preventing their successful adaptation to EHV's.

(B) Document the elimination process and submit the ECS elements Elimination List to JPL for review and approval.

(6) Task 6 - Ranking of ECS Elements

(A) Rank the ECS elements identified in Task (a)(3) individually.

(i) Use the JPL approved Rating Scheme developed and approved pursuant to Task (a)(2) activities.

(ii) In computing the rating scores use the following formula:

$$\text{Score} = \frac{W_1 S_1 + W_2 S_2 + \ldots + Wn\, Sn}{W_1 + W_2 + \ldots + Wn}$$

Where Si - the score given for the i^{th} rating criteria ($0 \leq Si \leq 100$)

and Wi - the weighting factor assigned to i^{th} rating criteria which defines its relative importance ($Wi > 0$)

(B) Document the ranking of each element considered.

(7) Task 7 - Identification of the "Best" ECS's

(A) Identify the "best" ECS for Electric Vehicles using state-of-the-art technology for possible immediate prototype development.

(B) Identify the "best" ECS for Electric Vehicles using innovative ECS element and/or subsystem concepts which may require more extensive development.

(i) For (a)(7)(A) and (B) above, use subsystem integration techniques and incorporate the relative vehicle driving range penalty computation as defined in (a)(7)(B)(i)b.

a. The relative driving range penalty shall be computed and applied as a final selection factor only to those ECS candidates which rank highest before application of this factor.

b. The relative vehicle driving range penalty shall be computed using the formula

$$\Delta R/R_o = \alpha_1 \Delta M + \alpha_2 \Delta D + \alpha_3 \Delta P$$

whereby ΔR is the change in vehicle range due to the ECS, Ro is the vehicle range before addition of the ECS, ΔM is the change in vehicle mass due to the ECS, ΔD is the change in product of aerodynamic drag coefficient and vehicle cross-section area due to the ECS, and ΔP is the average power required by the ECS not including power derivable from onboard fuel, if any.

c. Use the following sensitivity coefficients corresponding to Exhibit I, Attachment E, JPL version of SAE (Society of Automotive Engineers) J227a - Schedule D driving cycle:

1. α_1 = - .00026/lbm

2. α_2 = - .046/ft^2

3. α_3 = - .147/kW

(C) Document the "best" ECS selection process.

(8) <u>Task 8 - Impact of Adding Requirement for Battery Temperature Controller Cooling Capability</u>

(A) Since excessive electrolyte temperature may reduce cycle life of the battery pack, estimate the relative vehicle driving range penalty ($\Delta R/Ro$) if the ECS were required to provide a cooling capability to prevent battery electrolyte temperature from exceeding 100°F when the ambient temperature is a constant 120°F.

(B) Describe the required ECS modifications and/or additions needed to implement the cooling capability alluded to in paragraph (a)(8)(A) above.

(9) <u>Task 9 - "Best" ECS's Re-Assessment</u>

(A) Reassess the conclusion in Task (a)(7) considering a compact size Hybrid Vehicle.

(i) The Hybrid Vehicle shall have the following characteristics.

a. It shall be designed for 5 passengers and extended range use (approximately 250 miles between fueling stops).

b. The battery pack shall be assumed to be identical to the Electric Vehicle battery pack.

c. Initial charging of the batteries will be from house power and charging during operation will be from intermittent operation of a heat engine and generator (approximately 25HP).

(10) Task 10 - Estimates for ECS Prototype Development

(A) Generate cost, schedule, and confidence estimates for the development, fabrication, and testing of the "best" ECS systems identified in Tasks (a)(7) and (a)(9).

(11) Task 11 - Documentation and Briefings

(A) Contract Plans and Documentation

Prepare and submit all plans and documentation as defined by Contract Plans and Documentation, Exhibit II. Exhibit II consists of:

(i) Contract Data Requirements List (CDRL) which identifies the items to be delivered when delivery is required, the quantity and the type of each item and the frequency of issue.

(ii) Data Requirements Description (DRD) describes the basic documentation to be delivered.

(B) JPL/Contractor Meetings

Conduct a Final Review Meeting at JPL covering the results of the work performed under the Contract.

(b) The following documents are incorporated into and made a part of this Contract:

(1) Exhibit I - Applicable Documents List, dated June 15, 1979.

(2) Exhibit II - Contract Plans and Documentation, dated June 15, 1979.

(c) JPL will:

(1) Review and approve the Contractor documentation within ten (10) working days after the receipt of such documentation.

(2) Assist the Contractor in determining the (ΔD) product of aerodynamic drag coefficient and vehicle cross section area

APPENDIX B

EXHIBIT: ERICSSON-ERICSSON CYCLE HEAT PUMP/REFRIGERATOR DESCRIPTION (Reference 4)

3.2.2.2 Ericsson-Ericsson Cycle

An Ericsson cycle heat pump, driven by an Ericsson cycle heat engine is shown schematically in Figures 3-26a and 3-26b.

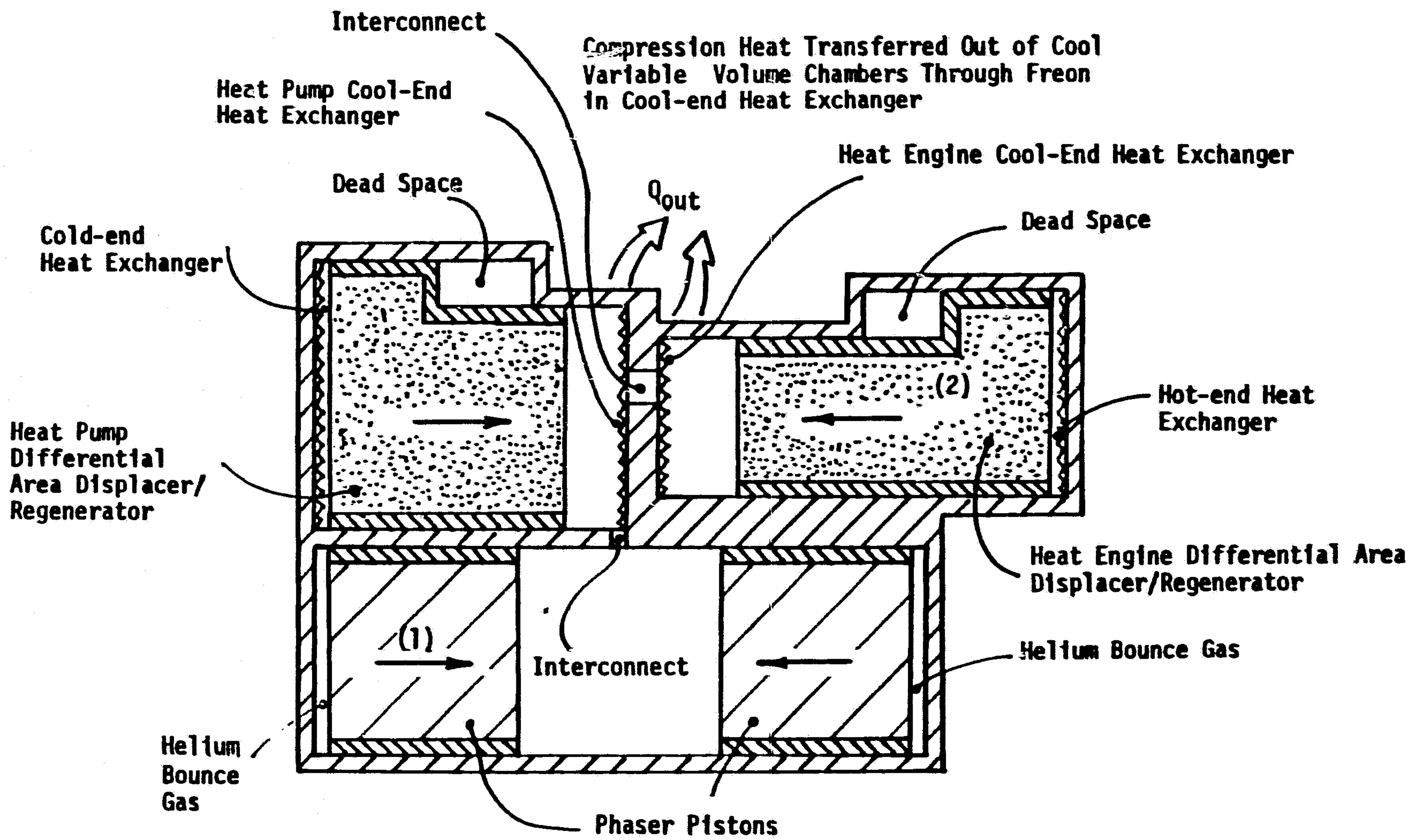
The Ericsson engine accepts heat from a hot source, converting a portion of this heat to mechanical work (which is absorbed by the heat pump), and rejecting the remaining heat to the environment. The operation of this cycle utilizes a heat pump regenerator/displacer, an engine displacer, phaser pistons, helium working gas, and freon heat transfer fluid. Both the displacer in the heat pump and in the engine are free piston displacers, and operate at steady state to maintain a constant volume of the combined heat pump and engine chambers, in opposition to the movement of the power pistons. This requires a differential area of the displacers; the cross-sectional area of the heat pump displacer/regenerator is larger at the cold end than at the cool end. The cross-sectional area of the heat pump chamber itself is larger at the cold end than at the cool end. Thus when the displacer/regenerator moves from the cold end to the cool end, the total heat pump chamber volume increases. When the displacer/regenerator moves from the cool end to the cold end, total heat pump chamber volume decreases.

Similarly, the cross-sectional area of the engine displacer is larger at the hot end than at the cool end, and the cross-sectional area of the engine chamber itself is larger at the hot end than at the cool end.

Thus, when the engine displacer moves from the hot end to the cool end, the total engine chamber volume increases. When the engine displacer moves from the cool end to the hot end, total engine chamber volume decreases.

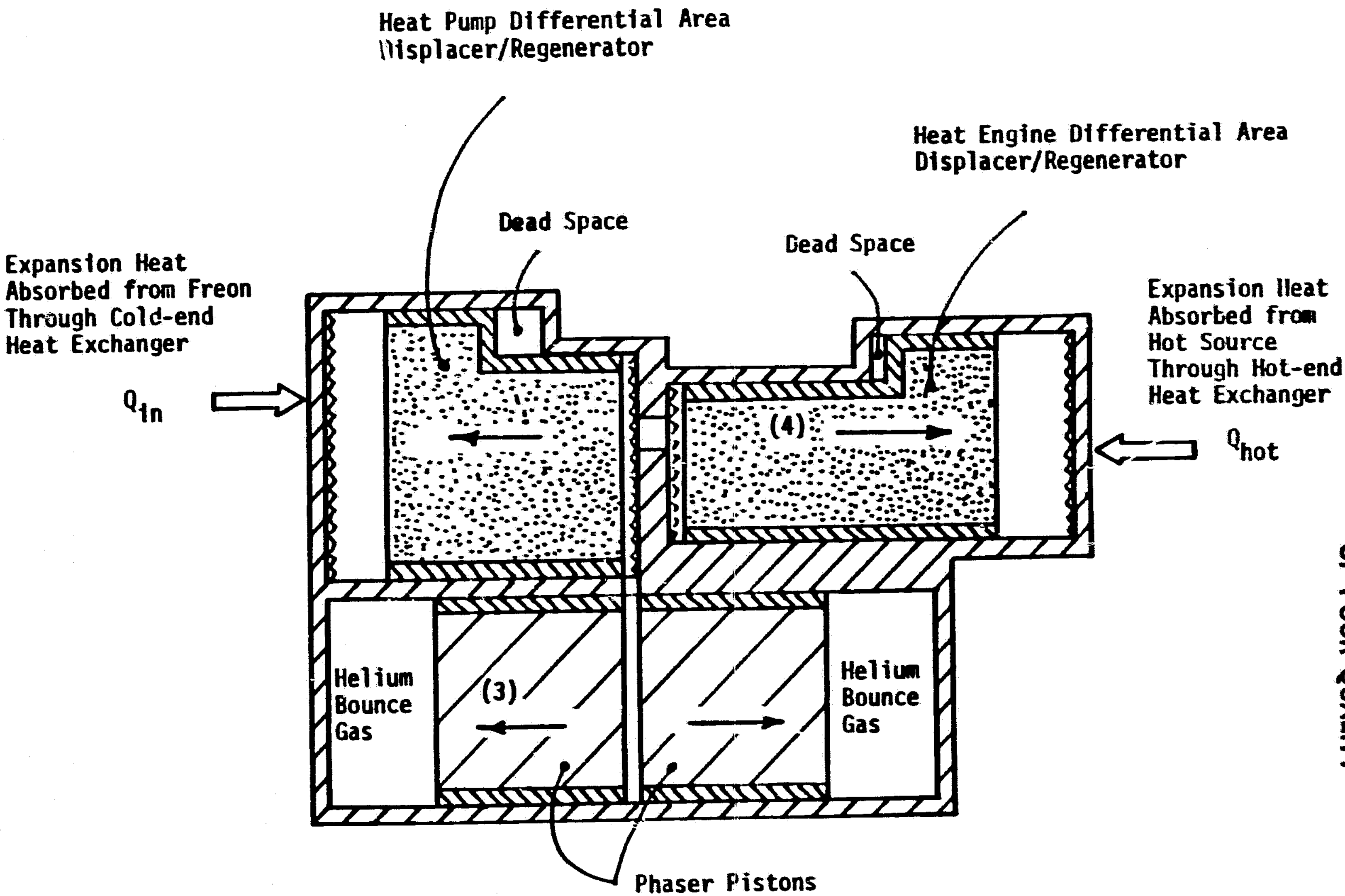
During start-up, the following steps occur:

- Heat is taken from the hot source (a burner, a thermal store, or a simple combustion heater) into the hot end of the engine. The temperature and pressure of the helium gas in the hot end increase.
- The increased pressure drives the engine displacer away from the hot end towards the cool end, transferring more Helium to the hot end, and thereby further increasing the Helium temperature and pressure.
- The increased pressure now causes the phaser pistons to stroke outward, expanding the Helium working gas, and compressing the Helium bounce gas in the phaser pistons (Figure 3-26a).



Source: Reference 30

Figure 3-26a. Ericsson-Ericsson Cycle Schematic - Compression and Regeneration - Steady State



Source: Reference 30

Figure 3-26b. Ericsson-Ericsson Cycle Schematic - Heat Addition and Regeneration - Steady State

- o Expansion of the Helium working gas lowers the total heat pump and heat engine chamber pressure. The displacers therefore move to decrease total chamber volume; the engine displacer returns to the hot end of the engine and the heat pump displacer/regenerator moves toward the cold end of the heat pump chamber.

- o The gas springs rebound the flywheel pistons, compressing the Helium working gas. As total chamber pressure is increased and the displacer pistons move to increase total chamber volume, the engine displacer moves toward the cool end of the engine, and the heat pump displacer/regenerator moves toward the cool end of the heat pump.

These oscillations build up with successive strokes, the heat pump displacer reciprocating in stroke opposition to the engine displacer.

In steady state operation, the Helium undergoes the following processes.

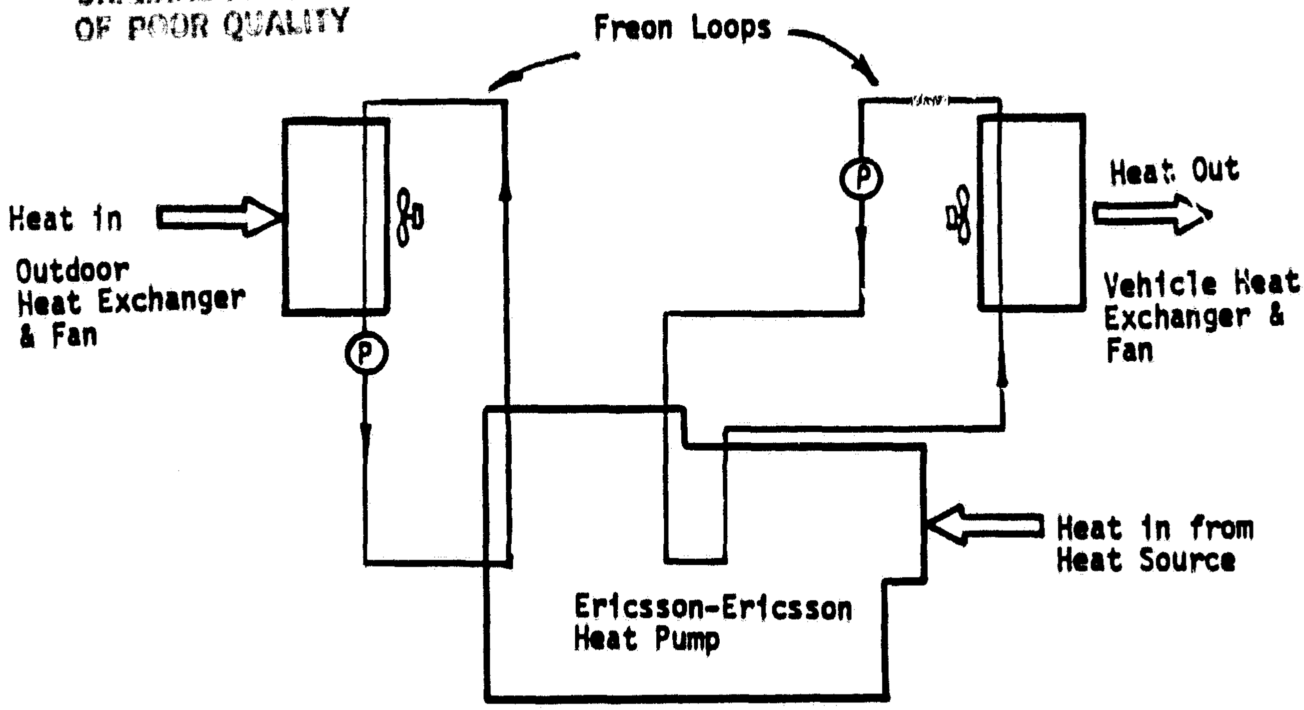
- o The phaser pistons move inward (1), thereby compressing the Helium working gas and raising its temperature. The heat of compression of the Helium is transferred to the freon at the heat pump cool-end and at the engine cool-end heat exchangers (Figure 3-26a).

- o When the power pistons compress the Helium, total pressure in the combined heat pump and engine chamberss increases. The displacers move to relieve the total pressure by increasing total chamber volume (2), the heat pump displacer/regenerator moves from the cold end toward the heat pump cool end, and the engine displacer moves from the hot end toward the engine cool end (Figure 3-26a).

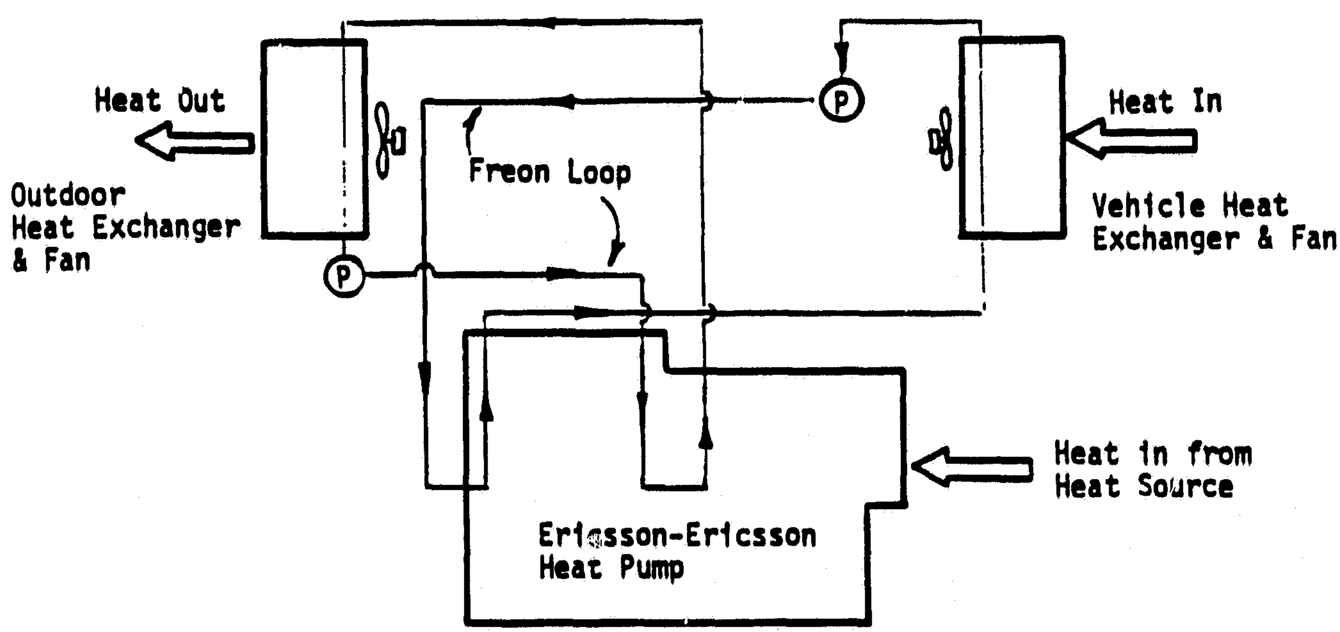
- o The phaser pistons move outward (3), expanding the Helium working gas. The expansion allows absorption of heat from freon in the heat pump cold end, and from the heat source in the engine hot end (Figure 3-26b).

- o With expansion, the total chamber pressure is lowered. The displacers therefore move to decrease the total chamber volume (4). The heat pump displacer/regenerator moves from the cool end towards the cold end and engine displacer moves from engine cool end towards the hot end (Figure 3-26b). The cycle then repeats.

For the vehicle heating service, the compression heat of Helium is absorbed by the freon in the heat pump cool end and in the engine cool end. Freon is transferred to the vehicle where it condenses, releasing heat to the passenger compartment (Figure 3-26c). For vehicle cooling, heat is absorbed from the passenger compartment by freon in the vehicle heat exchanger, which acts as an evaporator. The freon is transferred to the heat pump cold end, where it condenses, releasing heat to the expanding Helium (Figure 3-26c).



Heating Mode



Cooling Mode

P-Pump

Figure 3-26c. Ericsson-Ericsson Cycle Schematic - Heating and Cooling Mode

This system is designed so that, under startup or increasing heat pump load, the engine displacer advances relative to the heat pump displacer. The phaser stroke also increases. Together these produce a greater output per cycle of the machine. As the steady-state condition is approached, the displacers and phasers gradually phase shift to the steady state phase angles and increase their stroke to match the new load. As a result, the heat pump capacity increases with decreasing ambient temperature to meet the increased heating load required at lower ambient temperatures.

This system is still in the testing stage. The expected system performance indicates high efficiencies, at least 80 to 90 percent of the Carnot COP (90 to 95% Carnot efficiency each for the engine and the heat pump). Predicted COPs are shown in Figure 3-27 (Reference 21). Potentially, the production cost would be as low as that for a conventional electric motor driven, vapor compression unit of equal capacity, i.e., $1,100 for a 25,500 Btu/hr cooling capacity unit.

The physical characteristics for the engine/heat pump are given as 18 lbs and approximately 0.3 ft^3. Heat exchangers, fans, pumps, and a heat source will also be required (Figure 3-26c). A typical heat exchanger for a 25,500 Btu/hr cooling capacity unit weighs 20 lbs, and will add 1.0 ft^3 to the system (Reference 17). A typical fan weighs 7.5 lbs, and has a volume of 0.15 ft^3 (Reference 17). A typical pump for the Freon circulation weighs 9 lbs, and will add 0.12 ft^3 to the system (Reference 17). A heat source, in this case a simple combustion heater, will add 17 lbs and 0.8 ft^3 (Figures 3-3, 3-4). Thus total weight and size of the Ericsson-Ericsson heat pumps are 108 lbs and 3.6 ft^3.